International Journal of Mechanical and Production Engineering Research and Development (IJMPERD)
ISSN (P): 2249–6890; ISSN (E): 2249–8001
Vol. 10, Issue 2, Apr 2020, 605-614
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DETERMINATION OF FATIGUE ANALYSIS OF COMPRESSOR DISC IN GAS TURBINE

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ABSTRACT

An abaqus software is used to evaluate fatigue testing of tie bolt holes in a typical gas turbine compressor disc. The material properties needed for the disk are taken as a grade TI-8AL-1MO-1V Titanium alloy. The research focus is on the impact of Uniaxial loading and cyclic rotation speed is subjected to the disc. A solution is fully obtained under uniaxial loading for a disc, numerical analysis measures the problems for the disc.

KEYWORDS: Rotating Disc, Uni-Axial Stresses & Fatigue Test

Received: Jan 11, 2020; **Accepted:** Jan 31, 2020; **Published:** Mar 19, 2020; **Paper Id.:** IJMPERDAPR202060

Original Article

1. INTRODUCTION

They are normally tested under replicated engine orders, mainly in a spin rig setting, before gas turbine engine components are unconfined into operation. This test is carried out during the development stage of the part and during the validation and enhancement stages of the development process of the gas turbine engine. In the context of a life extension analysis, the spin rig test could also be acheived to evaluate the total fatigue life of engine components. Because of the reason that the cost of a successful life expansion test is much lower than the expense of a new aircraft, life extension studies using the "risk tolerance" approach are now conducted for aircraft. The damage tolerance approach to engine maintenance policies allows for components to be regularly inspected and undamaged parts to be returned to service for a specific number of airlift periods. Conserving retirement engines manages to significant investment for aircraft operators.

Historically, methods used to predict the life of components of gas turbine engine rotor have resulted in a conservative estimate of the useful life.' After achieving some predetermined life limit, components are withdrawn from service [2].

2. BACKGROUND OF A PROBLEM

The purpose of this paper is to investigate the propagation of fatigue crack and the effect of biaxial stress inherent in a rotating disc. Also measured are the stress strength factors of a minor crack formation from the bolt hole, or partially immersed in the plastic zone of the bolt hole [3].

3. METHODOLOGY

3.1 Spin Rig Test Facility

Spin checking is a significant step in stopping centrifugal catastrophe from bursting. Rotating components are used in an excessive – velocity. Turbine are working under great centrifugal loads and may fail with instable force. Therefore, all designers and manufacturers of turbo machine parts need to check the centrifugal power of the rotors.

Elevated speed disks are also essential to be proved by manufacturers before shipping and installation.

Figure 1: Spin Rig Facility.

The inner part of the rig chamber is enclosed with solid bricks made of lead to withstand the effect of turbine parts. Figure shows the machine chamber with turbine ready for investigation.

Figure 2: Image of Rig Chamber.

The study described here was performed as a preliminary examination before performing the turbine test of the laboratory spin rig. For that reason, this work does not contain the results of the laboratory turbine spin test.

3.2 Spin Rig Test and Fractographic Analysis

The rotating rig was driven by an air turbine which operated at constant repeated rotational speeds (8000—37,000—8000 rpm). The experiment was conducted at an atmospheric temperature, to reduce aerodynamic drag in air at low pressure.

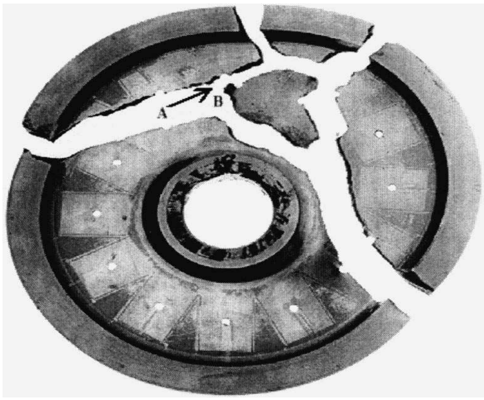

Figure 3: Broken Disc in Spin Rig Test.

Figure 4 shows a diagram of fault surface from hole A to hole B. The through-depth crack then quickly disseminated from hole A to hole B, and eventually the disc collapsed at 9470 cycles.

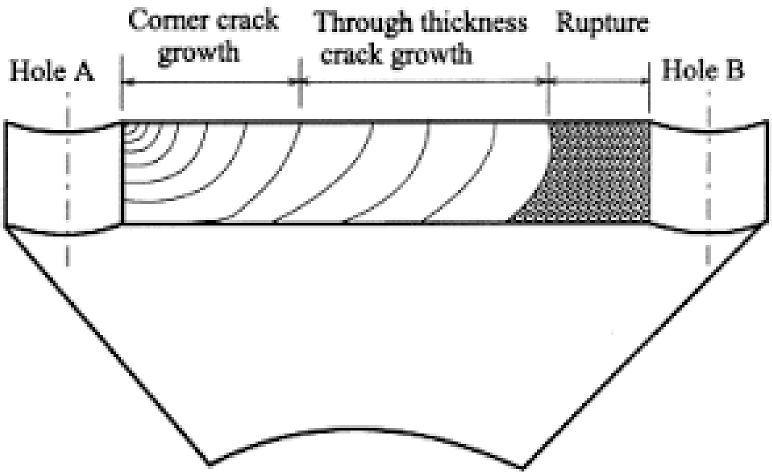

Figure 4: Fracture Surface between the Hole A and the Hole B in Broken Disc.

4. PROBLEM STATEMENT

The problem involves the fatigue study of a compressor disc in gas turbines. The parameters to be found are factor in stress intensity and growth rate of fatigue crack using J-integral approach b.

4.1 Finite Element Analysis (FEA) and Bolt Hole Stress State

The mesh of FEA is shown in Figure 5(a). All dimensions are in millimetres. The width of the bolt holes in the area is 3mm, with a thickness of 15mm at the hub and bottom. The mesh is composed of tridimensional quadratic brick elements of 1248 20-node. The range of cyclic speed used for the disk was 8500—37,500—8000 rpm. The structure in the region of the bolt hole is subjected to both outward and hoop stresses as shown in Figure 5(b).

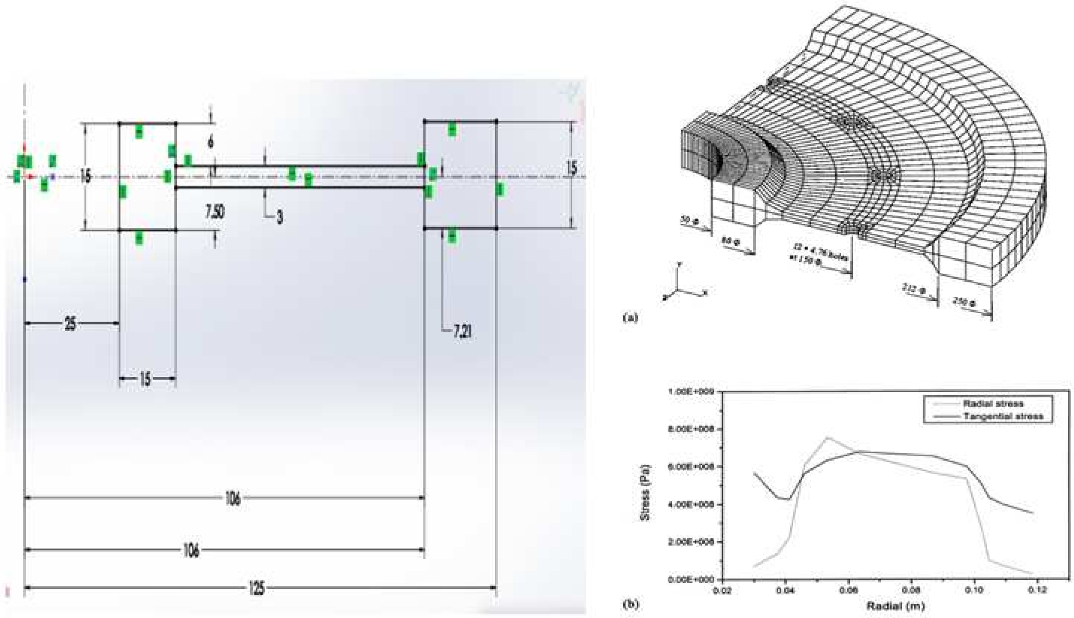

Figure 5: FE Stress Analysis of the Disc. (a). Disc Geometry and 3D FEmesh; (b) Distributions of outward and hoopstresses

Duplex annealed Ti–8Al–1Mo–1V titanium alloy was the disc material used for the spin rig test. Its ultimate tensile stress su is 1049 MPa, and the yield stress sy is 922 MPa (0.2 % offset). The elastic modulus E is 126.5GPa. The Poisson's ratio n is 0.32 and the density is 4372.9 kg/m^3.

From Figure .5(b), the stress condition in the region of the bolt hole is roughly constant. Therefore it is not unreasonable to assume that the outward and hoop stresses are distributed equally along the boundary, as shown in Fig.6. That will dramatically simplify the successive analysis of blow production.

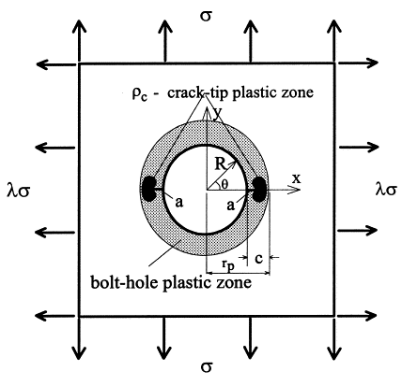

Figure 6: Stress Condition of bolt Hole.

The results of the finite element show that the stress state in the region of the bolt hole is approximately equi-biaxial. The radial tension at a minimum rotation speed of 8500 rpm is equivalent to 35 MPa with the same π. This means that the repeated charge stress ratio is about 0.045. The repeated load applied to the area of the bolt hole is shown in Figure.7

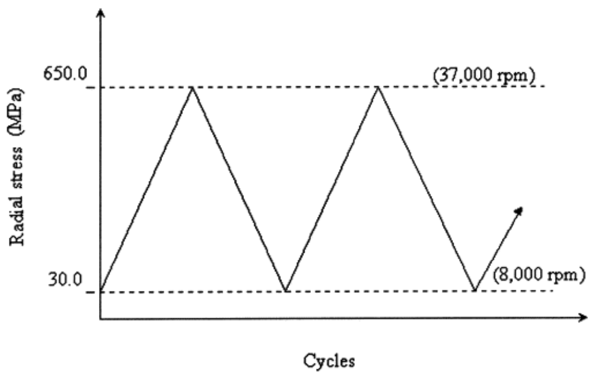

Figure 7: Radial Stress History for the bolt Hole Region.

The cumulative strain can be measured using the rule of Neuber, assuming a state of air stress and an elastic-perfect plastic material,

$$\varepsilon_{total} = \frac{(K_t \sigma)^2}{\sigma_{ys} E} = 0.0145 \quad (1)$$

Where Kt at the point of interest denotes stress concentration (in this case, Kt=2.0). Accordingly, the cumulative strain is determined to be approximately double the yield strain of the substance π0=ÿys / E=0.0073. However, since the stress applied on the farfield is repeated load with a stress ratio of 0.046.

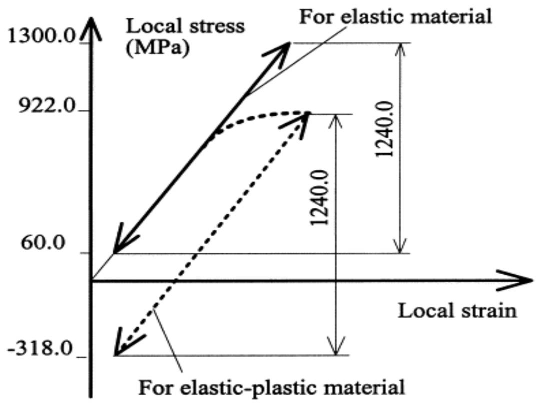

Figure 8: Repeated Load–Strain Response of bolt Hole.

5. ANALYSIS OF DISC

5.1 Introduction to ABAQUS (ABAQ)

Abaqus is a suite of effective simulation engineering programs focused on the finite-element approach provided by Dassault systems as part of their software tools for SIMULIA product life-cycle management (PLM). The MANE 4240/CILV4240 lectures will cover the basics of linear finite-element analysis with examples primarily from linear elasticity. The unique characteristics of abaqus include:

- ABAQ includes a large library of elements capable of modeling virtually any geometry.
- ABAQ can be designed as a general purpose simulation method to study more than just structural (stress / displacement) problems.
- ABAQ provides a wide array of simulation capabilities for linear and non-linear applications.

6. RESULTS AND DISCUSSIONS

The results for static structural analysis are done in Ansys workbench 18.0. So, the given experimental values are compared with these analysis values so that the values are to be accurate.

Table 1: Comparison of Experimental and Analysis Values

S. NO	Stresses	RPM	Analysis Values
1	Von-mises stress	At 37000rpm	652.82 MPa
2	Von-mises stress	At 8000rpm	32.52 MPa
3	Maximum stress	At 37000rpm	297.5 MPa
4	Maximum stress	At 8000rpm	147.6 MPa

6.1 Disc Calculations

Alternating stress (13.25-0/2) = 6.625

Mean stress (13.25/2) =6.625

Equivalent stress (1000*6.625/1049*6.625) =6.667

Failure cycles ln(6.67/2.786)/(0.0157) =4.95*10

Vonmises stress at 37000rpm=656.7 MPa

Vonmises stress at 8000rpm=29.67 MPa

Maximum principal stress at 37000rpm=293.0 Mpa

Maximum principal stress at 8000rpm=13.25 Mpa

Alternate stress = (max-min/2) = (293-0/2) = 146.5 Mpa

Mean stress = (max+min/2) = (293+0/2) = 146.5 Mpa

Equivalent stress = (UTS*alternate stress/UTS-Mean stress) = (1049*146.5/1049-146.5) = 170.280 Mpa

Failure cycles = e ^ln(σequ/x-coeff)/x- exponent = e^ln(170.2808/2786.2)/-0.157=5388+684.51

6.1.1 Explicit Dynamics

a Von-mises stress

Max stress

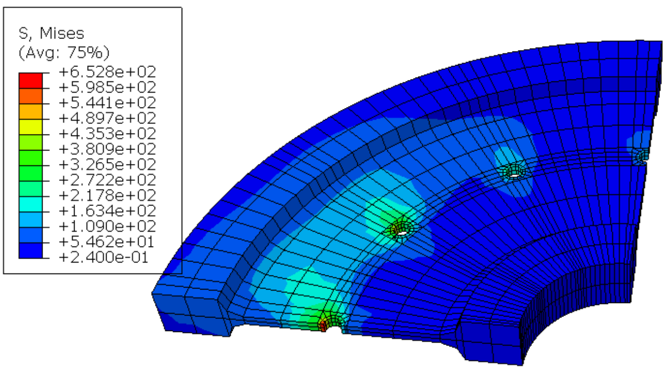

Figure 9: Vonmises Stress at 37000rpm.

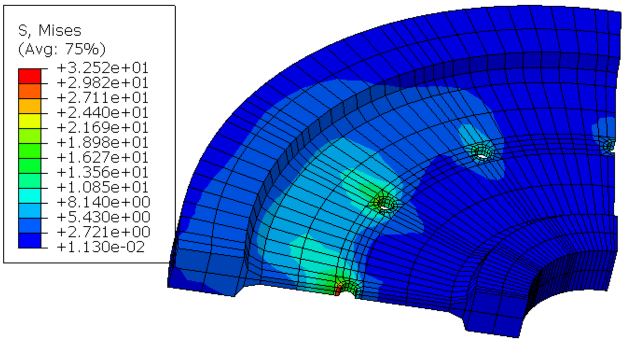

Figure 10: Von mises Stress at 8000rpm.

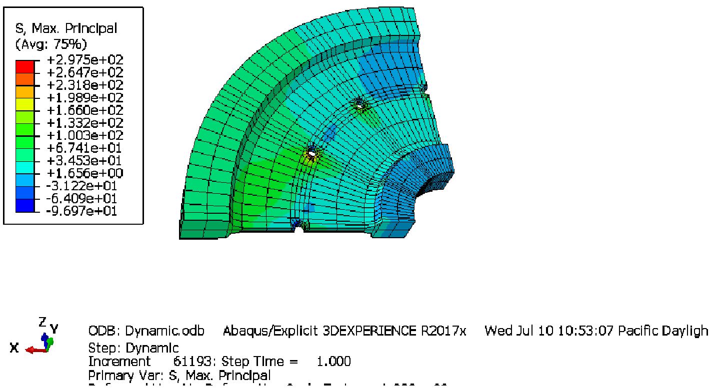

Figure 11: Maximum Stress at 37000rpm.

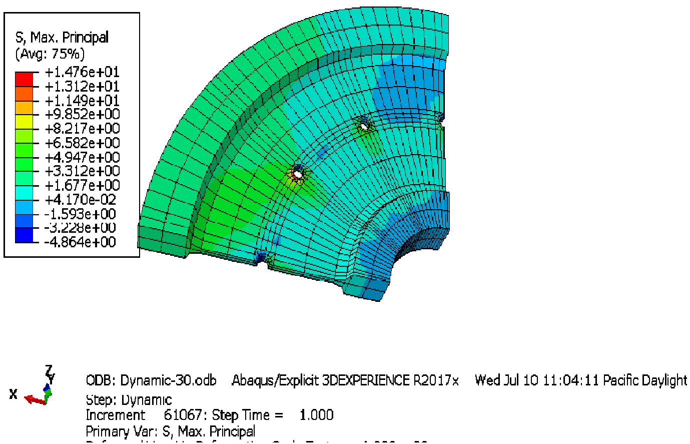

Figure 12: Vonmises Stress for 8000rpm.

6.1.2 Static Sructural

a Von-mises stress

Max stress

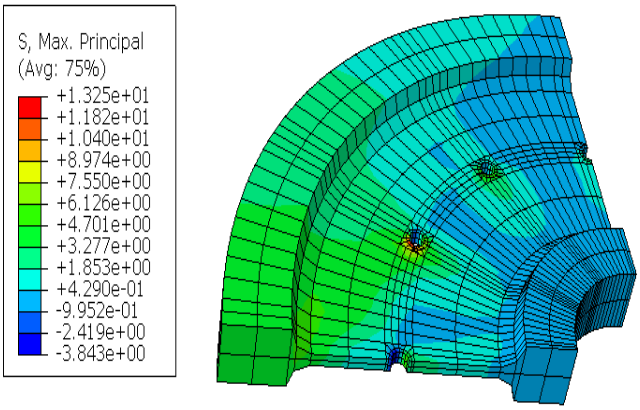

Figure 13: Maximum Stress at 8000rpm.

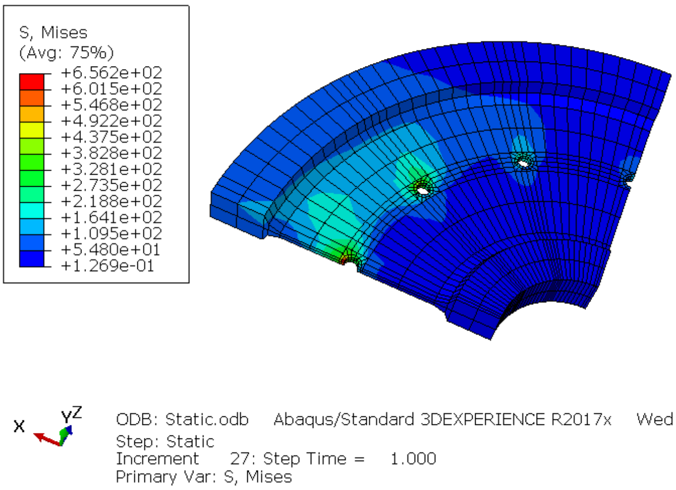

Figure 14: Vonmises Stress at 37000 rpm.

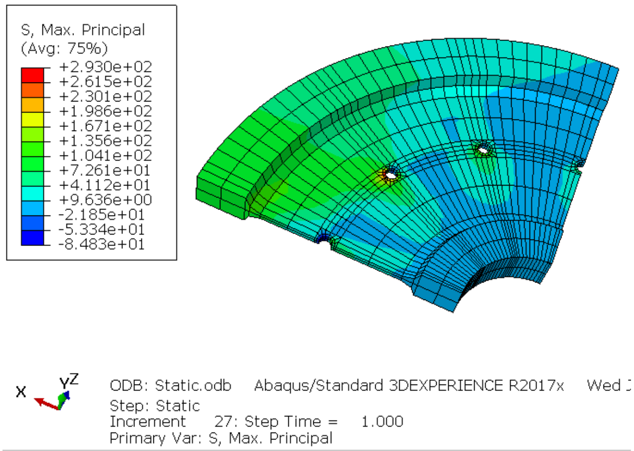

Figure 15: Max Principal Stress at 37000rpm.

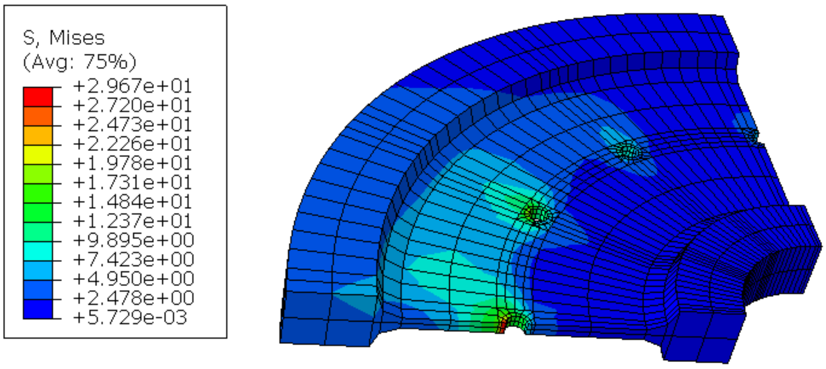

Figure 16: Von-mises stress at 8000rpm.

7. CONCLUSIONS

This study highlights number of matters that need to be tackled if we are to achieve better prediction.

Some strategies would be to use stress strength that influences along the blow forehead for a junction blow emanating from a circular hole under biaxial loading. The dissimilarity of the restraint factor along the blow face needs to be studied, particularly throughout the conversion from a corner crack to a through-thickness crack, and needed analysis is done.

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